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MULTI-CHANNELED LOOP HEAT TRANSFER DEVICE WITH HIGH EFFICIENCY FINS

BACKGROUND OF THE INVENTION

Field of the Invention

The present invention relates, in general, to a multi-channeled loop heat transfer device with high efficiency fins, used for transporting heat through phase change of working fluid within one loop and, more particularly, to a structural improvement in such a multi-channeled loop heat transfer device to form both an evaporating section and a condensing section using a plurality of parallel, multi-channeled flattened tubes and to optimally design the shape of the fluid channels within the flattened tubes, thus accomplishing both an effective capillary pumping operation and a smooth vapor feeding operation, the structural improvement of this invention being also to optimally design the shape of the outside fins of the device, thus forming active vortices and breaking the boundary layer growth so as to increase the heat transfer coefficient in the outside fins, which have high thermal resistance or bottleneck in heat transfer paths, and thereby improving the heat transfer rate of the device.

Description of the Prior Art

As well known to those skilled in the art, there are several thermal transport options capable of achieving thermal transportation between two spaced locations. These include conduction heat transfer, single-phase flow heat transfer, and two-phase flow heat transfer.

Since conventional conduction heat transport systems are substantially heavy and somewhat expensive, they have been limited only to some special applications.

The heat transport capacity of conventional single-phase heat transport systems is substantially larger than that of conventional conduction heat transport systems.

However, such a single-phase heat transport system necessarily requires an external power source for driving its mechanical elements, such as a pump, and so the device has not been preferably used for some applications requiring sensitive temperature control and a reduction in operation and maintenance cost. The single-phase heat transport system is also designed to perform sensible heat transport using the working fluid in a single phase, thus generating an undesirably large temperature drop. In order to overcome such a problem, it is necessary to dramatically increase the size of the heat exchanging section of the device.

On the other hand, conventional two-phase heat transport systems maintain an isothermal condition while transporting heat and require a low mass flow rate, and so it is possible to substantially reduce specific weight. Therefore, such two-phase heat transport systems have been preferably used for aerospace applications.

In such conventional heat transport systems, heat is transported from one location to another location through evaporation, condensation and mass flow.

Loop heat transfer devices use a thermodynamic cycle completely consuming work as pumping energy within one loop.

Such a thermodynamic cycle pumps working fluid up to an

evaporating section using the capillary pressure of a wick structure. The liquid working fluid absorbs sensible heat from the evaporating section and further absorbs latent heat, thus being finally evaporated.

The vaporization is created at the interface between the liquid and vapor phases on the capillary wick. During such vaporization, the working fluid expands in volume prior to flowing into the condensing section having the lowest vapor pressure.

In such a case, the pressure difference between the evaporating section and the condensing section is caused by the temperature difference between the vapor in the evaporating section and the vapor in the condensing section. Such a pressure difference acts as a practical driving force for many kinds of heat transfer devices.

In addition, thermal energy is transferred from the surroundings to the working fluid at the evaporating section, and is transported to the condensing section. At the condensing section, the working fluid is condensed while dissipating latent heat of vaporization to the surroundings.

The liquid tube of the loop heat transfer devices dissipates sensible heat to the surroundings, and so the liquid working fluid within the liquid tube is subcooled while maintaining a desired temperature difference at the interface between the liquid and vapor phases within the evaporating section.

Such conventional two-phase heat transport systems are classified into LHP (loop heat pipe) systems and CPL (capillary pumped loop) systems.

In the conventional LHP systems, both the evaporating section and the condensing section form a loop of a single-channeled

structure, thus having low heat transfer efficiency. In addition, the evaporating section of the LHP systems has a compensation cavity structure, and so the LHP systems undesirably have low response relative to load variation.

In the conventional CPL devices, both the evaporating section and the condensing section form a loop having a multi-channeled structure, and so the CPL devices have high heat transfer efficiency. However, the CPL devices also undesirably have a structural defect as follows: That is, the CPL devices require both a separate fluid reservoir for feeding working fluid to the wick structure within the evaporating section and a separate temperature regulator for controlling the liquid level since the evaporating section has a structural defect.

Furthermore, since the original design of both the LHP devices and the CPL devices is particularly for aerospace applications, they operate in the absence of fluid flow outside the pipes, and are suitable for radiant heat dissipation. Therefore, the two types of devices are problematic in that they provide only low heat transfer efficiency when used for general industrial or home applications.

Similar to most conventional heat pipe devices, both the conventional LHP devices and the conventional CPL devices require improved fin structure since the conventional fin structure undesirably forms high thermal resistance due to low outside heat transfer coefficient and forms a bottleneck in the heat transfer paths of the devices.

In the prior art, louver fins and offset strip fins (OSF) have been typically used as fins for heat exchanging sections. It

is noted that the louver fin is one of the highest in thermal efficiency of all kinds of conventional fins. However, such louver fins are problematic in that their thermal efficiency is undesirably reduced at positions around the rear strips of each fin. That is, since the main flow is formed along the louver at positions around the first half of each fin, the extent of the flow around the first half is lengthened. This finally accomplishes a desirably high heat transfer efficiency of the louver fins at positions around the initial part . However, since the extent of the main flow at positións around the rear parts of each fin is shortened, the heat transfer efficiency of the louver fins is regrettably reduced at positions around the rear parts . In addition, the strips of the conventional louver fins are designed to have an angle of attack of 90° with respect to the main flow. Therefore, the strips do not form any swirl in the flow, and so the strips fail to give high heat transfer efficiency, typically caused by longitudinal and transverse vortices , to the louver fins.

On the other hand, the conventional offset strip fins (OSF) are fabricated while being offset with each other, thus preferably breaking the boundary layer growth of the main flow due to a primary offset effect. However, such offset strip fins are problematic in that they fail to form any swirl in the main gas flow, thus merely providing a one-dimensional offset effect (unidirectional offset effect) along the main gas flow on the path line. Therefore, the heat transfer efficiency of the conventional offset strip fins is regrettably lower than that of the above-mentioned louver fins.

SUMMARY OF THE INVENTION

Accordingly, the present invention has been made keeping in mind the above problems occurring in the prior art, and an object of the present invention is to provide a multi-channeled loop heat transfer device with high efficiency fins, which forms each of the evaporating section and the condensing section using a plurality of parallel flattened tubes, and has a multi-channeled vapor flow path within the flattened tubes, and has a porous material wick, a matrix mat wick, a screen mesh wick, a curly woven wire wick in an axial direction, or an axially braided wire wick within each channel, and allows the wick to act as a fluid diode, and which also improves the wick structure so as to allow the wick to act as a desired fluid diode within a heat transport loop only when a minus temperature gradient is maintained in the liquid in the upstream of the boundary layer and in the vapor in the downstream of the boundary layer, and which accomplishes a desired temperature gradient at the evaporating section by supplying heat to opposite sides of the evaporating section, and which allows the loop to be operated at a subcooled condition with less temperature difference, and which separates the vapor flow path from the liquid flow path while allowing the temperature of the vapor flow path to be always maintained higher than that of the liquid flow path, thus allowing the liquid-phase working fluid to always flow from the condensing section into the liquid flow path due to vapor pressure of the vapor flow path, and which is thus preferably free from a fluid reservoir, a complex start unit, a control unit, a pump or other units typically required in starting and operating a conventional capillary pumped loop, and which is automatically restarted without failure when thermal load acts on the device

after a complete stop of the device.

The multi-channeled loop heat transfer device of this invention transports heat through a phase change of working fluid within a loop while accomplishing desired heat transfer efficiency higher than that of conventional heat pipes.

Since the heat transport action of a conventional heat transport system is performed by a phase change of working fluid, the thermal resistance of working fluid between the evaporating section and the condensing section of the device is negligible. Therefore, the thermal resistance, formed by external gas (atmospheric air), forms at least 90% of the total thermal resistance in the device. This finally means that the operational performance of the heat transfer device is almost completely determined by that of the fins.

It is thus possible to improve heat transfer efficiency of the heat transfer device by changing the geometrical shape of the fins and by reducing the thermal resistance of the fins.

A conventional structure having such improved heat transfer efficiency is referred to as an OLF heat exchanging structure, disclosed in Korean Patent Application No. 98-49,932 applied by the inventor of this invention. In this heat exchanging structure, the strips of each louver fin are designed to be oblique with an angle of attack β ($-90° \leq \beta \leq 90°$) relative to the main flow, thus forming transverse and longitudinal vortices in the main flow around the louver fin. Such vortices more violently mix the main flow and further improve the heat transfer efficiency of each louver fin. The oblique strips also force the main flow to collide with the flattened tubes acting as a base surface, thus improving the heat transfer rate of the flattened tubes. Each

oblique louver fin is closely corrugated from the first to the last end with the angle of attack β. Therefore, the louver fins preferably form a swirl in the main flow around the fins, thus providing a three-dimensional offset effect on the path line of the main flow. Such an oblique louver fin also lengthens the extent of the main flow around the fin. In addition, the flow length of the main flow around the louver fins may be controlled by changing the angle of attack β of the oblique strips of said louver fin, so that the louver fins more effectively and periodically break the boundary layer growth of the gas flow. That is, the oblique strips primarily reduce the thickness of the boundary layer and finally break the boundary layer growth within the vortex area defined between adjacent strips, thus improving the heat transfer rate of the louver fins. In a brief description, the oblique louver fins of the heat exchanging structure disclosed in the above Korean Patent Application are designed to have the advantages expected from both the typical louver fins and the typical offset strip fins. In addition, a desired number of flattened tubes are assembled into a module having a predetermined heat exchanging capacity. A desired number of modules are coupled into a single heat exchanging structure using a plurality of sockets, thus accomplishing a desired heat exchanging capacity of the resulting structure.

The present invention structurally improves the above-mentioned heat exchanging structure and uses the improved heat exchanging structure in a loop heat transfer device having two phases of fluid within one loop. The present invention thus provides a multi-channeled loop heat transfer device, of which both the evaporating section and the condensing section are formed

using a plurality of parallel and multi-channeled flattened tubes in place of conventional single-channeled circular pipes typically used in conventional loop heat transfer devices. In the device of this invention, the shape of the fluid channels within the flattened tubes is optimally designed, thus accomplishing both an effective capillary pumping operation and a smooth vapor feeding operation. In this invention, the shape of the outside fins is also optimally designed, thus forming active vortices within the device and breaking the boundary layer growth between two phases so as to increase the heat transfer coefficient in the outside fins, which have high thermal resistance or bottleneck in the heat transfer paths, and finally improving the heat transfer rate of the device.

In the device of this invention, each of the evaporating and condensing sections comprises a plurality of flattened tubes fitted into the fitting slits of two opposite headers, with two support frames extending at the top and bottom ends of the flattened tubes while supporting the flattened tubes and blocking both ends of the headers.

BRIEF DESCRIPTION OF THE DRAWINGS

The above and other objects, features and other advantages of the present invention will be more clearly understood from the following detailed description taken in conjunction with the accompanying drawings, in which:

Fig. 1a is a front view of an evaporating section included in the loop heat transfer device in accordance with the preferred embodiment of the present invention;

Fig. 1b is a plan view of the evaporating section of Fig. 1a, showing an upper header;

Fig. 1c is a bottom view of the evaporating section of Fig. 1a, showing a lower header;

Fig. 2a is a perspective view of a header included in the loop heat transfer device in accordance with the preferred embodiment of this invention;

Fig. 2b is a perspective view of a header in accordance with a modification of the preferred embodiment of this invention;

Fig. 3a is a view of the brazed junction between a header and a flattened tube within the evaporating section of this invention;

Fig. 3b is a sectional view of the brazed junction between the header and the flattened tube within the evaporating section of this invention;

Fig. 3c is a sectional view, showing a wick provided at the junction between a lower header and a flattened tube included in the loop heat transfer device of this invention;

Fig. 4a is a perspective view, showing the construction of the evaporating section with surface extension fins, the evaporating section having both an external condensate drainage slot and an internal liquid pumping H-shaped wick of this invention;

Fig. 4b is a perspective view, showing the construction of the evaporating section with surface extension fins, the evaporating section provided with both an external condensate drainage slot and an internal liquid pumping I-shaped strip wick of this invention;

Fig. 5a is a sectional view of the evaporating section having multi-channeled flattened tubes with interior division fins

according to this invention;

Fig. 5b is a side view of the outside surface extension fins brazed to the flattened tubes of this invention;

Fig. 5c is a development view of one of the surface extension fins of Fig. 5b;

Fig. 6 is a view, showing the construction a multi-channeled flattened tube with a brazed fin included in the loop heat transfer device of this invention;

Fig. 7a is a view, showing the construction of a multi-channeled flattened tube according to an embodiment of this invention;

Fig. 7b is a view, showing the construction of a multi-channeled flattened tube according to another embodiment of this invention;

Fig. 7c is a view, showing the construction of a multi-channeled flattened tube according to a further embodiment of this invention;

Fig. 7d is a view, showing the construction of a multi-channeled flattened tube according to a still another embodiment of this invention;

Fig. 7e is a view, showing the construction of backflow prevention liquid diodes used as lower end caps for the vapor flow path of the evaporating section included in the loop heat transfer device of this invention;

Figs. 8a to 8h are views, showing the shape of fins having an angle of attack in accordance with the preferred embodiments of this invention;

Fig. 9a is a front view of a two-row air reheating, vapor condensing section included in the loop heat transfer device in

accordance with this invention, showing the brazed junction between the headers and multi-channeled flattened tubes;

Fig. 9b is a plan view of the two-row air reheating, vapor condensing section included in the loop heat transfer device of this invention;

Fig. 9c is a bottom view of the two-row air reheating, vapor condensing section included in the loop heat transfer device of this invention;

Fig. 10 is a sectional view of a multichanneled flat tube with micro-fins or micro-grooves which enhance heat transfer in the loop heat transfer device of this invention;

Fig. 11a is a view of a portable dehumidifying and reheating type air conditioner using the loop heat transfer device of this invention;

Fig. 11b is a view of a portable dehumidifying and reheating type air conditioner using the loop heat transfer device in accordance with another embodiment of this invention;

Fig. 11c is a view, showing a portable dehumidifying and reheating type air conditioner using the loop heat transfer device in accordance with a further embodiment of this invention;

Fig. 12a is a view, showing a portable dehumidifying and reheating type cool air generator using the loop heat transfer device in accordance with still another embodiment of this invention;

Fig. 12b is a view, showing a portable dehumidifying and reheating type cool air generator using the loop heat transfer device in accordance with still another embodiment of this invention;

Fig. 12c is a view, showing a portable dehumidifying and

reheating type cool air generator using the loop heat transfer device in accordance with still another embodiment of this invention; and

Fig. 12d is a view, showing a portable dehumidifying and reheating type cool air generator using the loop heat transfer device in accordance with still another embodiment of this invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Fig. 1a is a front view of an evaporating section included in the loop heat transfer device in accordance with the preferred embodiment of the present invention. Fig. 1b is a plan view of the above evaporating section. Fig. 1c is a bottom view of the above evaporating section. Fig. 9a is a front view of a two-row air reheating, vapor condensing section included in the loop heat transfer device of this invention, showing the brazed junction between the headers and multi-channeled flattened tubes. Fig. 9b is a plan view of the two-row air reheating, vapor condensing section. Fig. 9c is a bottom view of the two-row air reheating, vapor condensing section.

As shown in the drawings, each of the evaporating section and the condensing section included in the loop heat transfer device according to the preferred embodiment of this invention comprises a plurality of fluid supply flattened tubes 11. The above flattened tubes 11 are fitted into the fitting slits of two opposite headers 13 at both ends thereof, thus extending between the two headers 13 while being regularly spaced and being parallel to each other. Each of the flattened tubes 11 is obliquely cut at

a cutting angle of 90° at its both ends and has an oblique louver fin on its top end. This louver fin is repeatedly and obliquely cut at an angle of attack of β on its surface, thus having a plurality of oblique strips. The above oblique strips allow a main gas flow to collide thereon when the gas flow passes by the louver fin. Two support frames 21 extend at the top and bottom ends of the flattened tubes 11 while supporting the flattened tubes 11 and blocking both ends of the headers 13 using a plurality of end caps 15. Each of the support frames 21 is longitudinally channeled, thus having a U-shaped cross-section. In the present invention, a desired number of flattened tubes 11 are assembled with two opposite header pipes 13 into a module having a predetermined heat exchanging capacity. A desired number of modules are coupled into a single structure using a plurality of sockets, thus accomplishing a desired size and capacity of a resulting section.

The side oblique angle γ of the above oblique louver fin is set to $-60° \leq \gamma \leq 60°$, while the angle of attack β of the louver fin is set to $-90° \leq \beta \leq 90°$.

In the preferred embodiment of this invention, the evaporating section is fabricated to have a single-row structure, while the condensing section is fabricated to have a one or more row structure (two-row is shown in the drawings Fig.9a, Fig.9b, Fig.9c for convenience) since the thermal capacity of the condensing section is typically larger than that of the evaporating section.

In the drawings, the reference numeral 16 denotes fluid inlet and outlet ports.

Fig. 2a is a perspective view of a header included in the

loop heat transfer device in accordance with the preferred embodiment of this invention. Fig. 2b is a perspective view of a header in accordance with a modification of the preferred embodiment of this invention. Fig. 3a is a view of the brazed junction between a header and a flattened tube within the evaporating section of this invention. Fig. 3b is a sectional view of the brazed junction between the header and the flattened tube of the evaporating section. Fig. 3c is a sectional view, showing a wick provided at the junction between a lower header and a flattened tube included in the loop heat transfer device of this invention. Fig. 4a is a perspective view, showing the construction of the evaporating section with outside extended surface fins , the evaporating section having both an external condensate drainage slot and an internal liquid pumping H-shaped wick of this invention. Fig. 4b is a perspective view, showing the construction of the evaporating section with outside extended surface fins, the evaporating section provided with both an external condensate drainage slot and an internal liquid pumping I-shaped strip wick of this invention. Fig. 5a is a sectional view of the evaporating section having multi-channeled flattened tubes with interior division fins according to this invention. Fig. 5b is a side view of the outside surface extension fins brazed to the flattened tubes of this invention. Fig. 5c is a development view of one of the surface extensions fin of Fig. 5b. Fig. 6 is a view, showing the construction a multi-channeled flattened tube with a brazed fin included in the loop heat transfer device of this invention.

As shown in the drawings, the ends of the flattened tubes 11 are inserted into the fitting slits 19 of the headers 13 and are

brazed to the headers 13 at the portions 22.

Each of the flattened tubes 11 is provided with multiple channels having a circular or polygonal shape, such as a rectangular shape, as shown in Figs. 7a, 7b, 7c and 7d. The individual channels of the flattened tubes 11 communicate with each other. A variety of wicks 17a, 17b and 17c are set within the channels and act as capillary pumps for liquid.

Fig. 3c shows a capillary pumping structure, in which liquid within the lower header is pumped up by the wicks set within the flattened tubes.

A central condensate drainage slot 23 is formed on the central portion of the external surface of each flattened tube, with a side condensate drainage slot 23a formed on each of opposite side portions of the external surface of each flattened tube.

On the other hand, a central condensate drainage hole 24 is formed at the central portion of each OLF fin.

Each of the headers 13 is provided with a plurality of fitting slits 19 and is coated with a melting material of aluminum brazing 13a.

The construction of the loop heat transfer device according to this invention will be described in more detail herein below.

The device of this invention comprises a plurality of multi-channeled flattened tubes, which are fitted into the fitting slits of two opposite headers at both ends thereof, thus extending between the two headers while being regularly spaced and being parallel to each other. Each of the flattened tubes has an oblique louver fin on its top end. This louver fin is repeatedly and obliquely cut at an angle of attack of β on its surface, thus

having a plurality of oblique strips.

The headers hold the flattened tubes within each of the evaporating and condensing sections. In addition, the number of fluid pumping wicks arrayed within the channels of the flattened tubes is controlled in accordance with a desired thermal capacity of the headers. It is thus possible to control the pressure loss and mass rate of vapor flow within the device as desired.

The evaporating section consists of a plurality of flattened tubes, which are fitted into the fitting slits of two opposite headers, with two support frames extending at the top and bottom ends of the flattened tubes while supporting the flattened tubes and blocking both ends of the headers. A plurality of end caps, made of a porous material, are set within all the lower portions of the evaporating section, thus preventing a backflow of vapor and acting as fluid diodes.

The device of this invention also has a condensing section. This condensing section has a construction similar to that of the above-mentioned evaporating section, but is free from any condensate drainage slot on its flattened tubes and free from any wick within the channels of the flattened tubes.

The loop heat transfer device of this invention also has a vapor pipe, which is thermally insulated so as to reduce thermal loss within the device. In the device of this invention, it is possible to reduce the outside diameter of the vapor pipe in comparison with a conventional heat pipe having the same thermal capacity. The thermal loss of this device is thus remarkably lower than that of conventional heat pipes. However, an increase in thermal loss at the insulated portion undesirably reduces thermal potential, and so it is necessary to sufficiently

thermally insulate the vapor pipe.

This loop heat transfer device also has a liquid pipe, which is sufficiently subcooled so as to prevent an undesired boiling at the inlet of the evaporating section.

In the device of this invention, the liquid supply wick have a function of wicks for preventing vapor backflow.

In this device, heat is dissipated from the external surfaces of the condensing section, and so the working fluid within the condensing section is desirably condensed. The condensed working fluid, thereafter, flows from the condensing section into the lower header of the evaporating section through the liquid pipe. Therefore, this liquid pipe acts as a liquid return pipe in the device.

In the present invention, it is possible to design the liquid pipe of the loop heat transfer device to have a serpentine structure capable of more effectively subcooling the working fluid using external condensate .

The vapor pipe and the liquid pipe of this device are separated from each other. In addition, the vapor pipe, having an outer diameter larger than that of the liquid pipe, is always maintained at a temperature higher than that of the liquid pipe, thus allowing the gaseous working fluid to always flow into the liquid pipe due to vapor pressure within the vapor pipe.

The device of this invention is a two-phase loop heat transfer device having wicks. When thermal load acts on the device after the device is completely stopped, this device is automatically restarted without allowing a backflow of fluid since the device has a backflow prevention means.

In the device of this invention, the condensing section,

liquid pipe and vapor pipe are free from a wick, while the evaporating section is provided with wicks.

Fig. 7a is a view, showing the construction of a multi-channeled flattened tube according to an embodiment of this invention. Fig. 7b is a view, showing the construction of a multi-channeled flattened tube according to another embodiment of this invention. Fig. 7c is a view, showing the construction of a multi-channeled flattened tube according to a further embodiment of this invention. Fig. 7d is a view, showing the construction of a multi-channeled flattened tube according to a still another embodiment of this invention. Fig. 7e is a view, showing the construction of liquid diodes for preventing backflow as lower end caps for the flow path of the evaporating section included in the loop heat transfer device of this invention.

In the evaporating section of this invention, the internal path of each flattened tube, which is divided by both a multi-channel wall 11a and a plurality of multi-channel division fins 11b, is provided with capillary pumping wicks 17 (17a, 17b, 17c), vapor flow paths 14, 14a and 14b, vapor lock prevention clearance 17d which eliminate vapor lock due to the contact between wall and wick, and backflow preventing fluid diode 17e.

In the present invention, a curly woven wire wick 17a twisted in an axial direction, a porous material wick 17b, or a mesh wick 17c may be used as the capillary pumping wick 17.

In the device of this invention, the evaporating section is formed by multi-channeled parallel flattened tubes in place of conventional circular pipes, with the vapor path of the flattened tubes having a circular, elliptical, square or rectangular cross-section.

In addition, the evaporating section is formed by multi-channeled flattened tubes connected to a loop heat pipe in parallel. The wall between the channels acts as a division wall, and so vapor flows into the wick-free path in the case of an application of high external thermal load. Therefore, it is possible to reduce the vapor flow pressure loss and to reduce the pressure difference between the vapor flow paths, and to improve the operational flexibility in response to variations in thermal load.

When the liquid phase working fluid is vaporized, specific weight of the working fluid is increased by several hundred times to one thousand or more times. Therefore, the number of capillary pumping wick-free vapor pipes is set to be larger than that of the vapor pipes having such wicks, thus accomplishing a desired balance between the liquid flow and the vapor flow.

In the case of an evaporating section made of vertically positioned flattened tubes, heat is absorbed from the outer surfaces of the evaporating section, with a capillary pumping structure formed as follows. That is, the hollow polygonal wick or the H-shaped wick set within each channel of the flattened tubes has a porous media wick structure, and so a desirable positive temperature gradient is maintained in the vapor flowing within the evaporating section. In this invention, the H-shaped wick is preferably set on the division wall.

In the case of an evaporating section made of horizontally positioned flattened tubes, heat is absorbed from the outer surfaces of the evaporating section, with a capillary pumping structure formed as follows. That is, the I-shaped strip wick set within each channel of the flattened tubes has a porous media wick

structure, and so a desirable positive temperature gradient is maintained in the vapor flowing within the evaporating section. The above I-shaped strip wick is preferably set on the division wall.

The axially braided micro wire wick has a structure twisted in an axial direction outside the spring which support wick in a radial direction.

On the other hand, the mesh wick is formed by a plurality of micro-wire woven screen meshes set within each channel to form a multi-layered wick structure.

In each of the above wick structures, an end cap, made of a porous material, is installed at all the lower ends of the vapor flow path in order to prevent an undesired vapor backflow at the liquid/vapor interface within the evaporating section regardless of an application of high external thermal load. The above end cap thus acts as a fluid diode within the heat transport loop.

Figs. 8a to 8h show the shapes of fins having an angle of attack of this invention. As shown in the drawings, it is possible to promote a fluid mixing effect due to a discrete effect formed by cutting the strips so as to give both an angle of attack β and a variety of shapes to the strips. This finally improves the outside heat transfer coefficient of the loop heat transfer device. Particularly, Fig. 8h shows a fin shape, of which the louver pitch is gradually increased or reduced in a downstream direction so as to accomplish a desired louver directed flow even when external air flows within the device of this invention at a low speed.

In the drawings, the reference numerals 12a to 12d denote OLF fin strips having a variety of shapes, 12e and 12f denote square,

rectangular, circular, elliptical or hexagonal perforated holes formed on each OLF fin, 12g denotes a bump formed on the OLF fin so as to accomplish a flow disturbance and to promote heat transfer from the fin, and 12h denotes irregularly spaced strips formed on the OLF fin so as to allow a boundary layer to be formed at a low speed flow section.

Fig. 10 is a sectional view, showing multi-channeled and micro-finned flattened tubes, which have division fins and are used for promoting the evaporation and condensation effect within the device of this invention. As shown in the drawing, each of the evaporating and condensing sections is formed using the flattened tubes connected to the loop heat pipe in parallel, with a multi-channel wall having a micro-grooved surface or a micro-finned surface.

The multi-channeled loop heat transfer device of this invention will be preferably used in thermal systems as follows.

Figs. 11a to 11c are views of portable dehumidifying and reheating type air conditioners using the above-mentioned loop heat transfer device in accordance with different embodiments of this invention. Each of the above air conditioners has a construction similar to that of a cool air generator. However, different from the cool air generator, each air conditioner of this invention has a small-sized air conditioning system. This air conditioning system comprises an evaporator installed in place of a cold storage box (with an ice capsule or a phase change material capsule) used in the cool air generator for reducing temperature. The air conditioning system also comprises a compressor , a condenser and an expansion valve, being used for feeding low temperature refrigerant to the evaporator.

In each of the air conditioners, the evaporator of the small-sized air conditioning system is installed at a position between the evaporating and condensing sections of the loop heat transfer device. In an operation of each of the air conditioners, moist air is firstly reduced in temperature while passing through the evaporating section and is secondarily reduced in temperature and absolute humidity while passing through the evaporator of the small-sized air conditioning system. The air is thus cooled and dehumidified so as to become lower temperature air. This air is increased in temperature and is reduced in relative humidity while passing through the condensing section of the loop heat transfer device and is secondarily increased in temperature while passing through the condenser of the small-sized air conditioning system, thus becoming air of a suitable temperature. This air is, thereafter, discharged into and mixed with room air by the blowing force of a motored fan. The air-conditioned space become comfortable.

Condensate from the air dehumidifying process of the air conditioner is drained to a water reservoir installed at the lower section of the air conditioner.

In the present invention, it should be understood that the arrangement of the evaporator and condenser of the small-sized air conditioning system and the evaporating and condensing sections of the heat transfer device may be appropriately changed as desired. Such an alteration of the arrangement is shown in Figs. 11a to 11c.

In addition, it is possible to appropriately control the operating conditions of the small-sized air conditioning system, such as the amount of refrigerant, as desired.

Figs. 12a to 12d are views, showing portable dehumidifying and reheating type cool air generators using the loop heat transfer device in accordance with different embodiments of this invention. Each of the cool air generators of this invention comprises an external case, an air suction port, an air discharge port, a blower fan, a motor, a cold storage box (with an ice capsule or a phase change material capsule) used for reducing temperature, a water reservoir for containing water from the air dehumidifying process, and the loop heat transfer device of this invention.

In each of the cool air generators, a plurality of cold storage boxes (with an ice capsule or a phase change material capsule) are installed at a position between the evaporating and condensing sections of the loop heat transfer device. In an operation of each of the cool air generators, moist air is firstly reduced in temperature while passing through the evaporating section of the loop heat transfer device and is secondarily reduced in temperature and absolute humidity while passing through the cold storage boxes ice capsules or phase change material capsules, thus becoming low temperature humidified air. This air is also reduced in relative humidity while passing through the condensing section and is discharged into a room by the blowing force of a motored fan.

In the case of the above cool air generators, condensate from the air dehumidifying process of the cool air generator is drained to the water reservoir installed at the lower section of the cool air generator.

In the present invention, it should be understood that the arrangement of the evaporating and condensing sections of the heat

transfer device may be appropriately changed as desired. Such an alteration of arrangement is shown in Figs. 12a to 12d.

As well known to those skilled in the art, conventional cool air generators are problematic as follows. Conventional cool air generators are designed to forced circulating air using the blowing force of a motored fan, thus allowing the air to pass through a wet pad and reducing the temperature of the air using latent heat formed by vaporization of water. However, they only provide cold and highly humid air, having a high relative humidity of not less than 90% due to vapor, to users, thus causing the users to feel sticky and uncomfortable with moisture on the skin. This finally reduces the market competitiveness of the conventional cool air generators. However, such a problem experienced in the conventional cool air generators can be almost completely overcome by the cool air generators using the loop heat transfer device of this invention.

Anyway, the loop heat transfer device of this invention may be preferably used in portable air conditioners, portable dehumidifiers, portable dryers, devices for recovering energy from exhaust gas, and white smoke preventing devices in the chimney.

As described above, the present invention provides a multi-channeled loop heat transfer device with high efficiency fins. This loop heat transfer device has a flattened tube structure different from a conventional heat pipe having the same hydraulic diameter, thus being remarkably reduced in flow pressure loss. The loop heat transfer device of this invention has high efficiency OLF fins.

Such a reduction in flow pressure loss preferably increases

the flow rate under the same condition of pressure drop. In the multi-channeled loop heat transfer device of this invention, such an increase in flow rate allows the heat transfer enhancement at the evaporating and condensing sections to be remarkably improved due to three-dimensional offset effect.

The conventional heat pipes have substantial thermal loss in the vapor and liquid flow at the evaporating section, insulating section and condensing section.

Such thermal loss undesirably reduces thermal potential, and so it is necessary to completely insulate the vapor pipe in order to allow the vapor pipe to perform a desired operational function. Since the vapor pipe within the heat transfer device of this invention is completely separated from the liquid pipe, it is possible to remarkably reduce such thermal loss.

In conventional heat pipes, a thermal conduction through a wall/wick structure within the heated evaporating section undesirably causes a temperature drop. Such a temperature drop is also made by the condensed working fluid flowing to the evaporating section in the conventional heat pipes. That is, the normal internal temperature drop, generated during a heat transfer process of a conventional heat pipe, is a fluid conduction drop. In addition, the heat flux, formed in the evaporating section of such a conventional heat pipe, is undesirably limited by a boiling at the wick/liquid interface.

However, in the loop heat transfer device consisting of parallel, multi-channeled flattened tubes of this invention, heat is directly transferred to the liquid/vapor interface through the walls of the flattened tubes and is also transferred to said interface through a convection effect formed by vapor flow. The

heat transfer device of this invention thus preferably reduces the temperature drop. In an operation of the loop heat transfer device of this invention, vapor and liquid separately flow in two separate pipes, thus increasing the amount of evaporated and condensed working fluid different from conventional heat pipes. This finally improves the evaporation and condensation effect of the heat transfer device and remarkably increases the amount of heat transferred thermodynamically .

In an operation of the conventional heat pipes, two phases flow in opposite directions within one pipe, while a liquid distribution within the pipe is affected by gravity . Therefore, the gravity effect deteriorates the operation performance of conventional heat pipes.

However, in the loop heat transfer device of this invention, the two phases of fluid separately flow in the same direction within separate pipes without being affected by the gravity . Therefore, the heat transfer device of this invention remarkably reduces the pressure drop and remarkably improves heat transport capacity.

In the loop heat transfer device of this invention, the evaporating section may be fabricated with vertically positioned flattened tubes or horizontally positioned flattened tubes. In the case of an evaporating section made of vertically positioned flattened tubes, an H-shaped wick, having a porous media wick structure, is held by the division wall within each channel of the flattened tubes, and so a desirable positive temperature gradient is maintained in the vapor flowing within the evaporating section. In the case of an evaporating section made of horizontally positioned flattened tubes, an I-shaped strip wick, having a

porous media wick structure, is held by the division wall within each channel of the flattened tubes, and so a desirable positive temperature gradient is maintained in the vapor flowing within the evaporating section. Therefore, the device of this invention accomplishes desired operation performance regardless of the installed direction of the flattened tubes within the evaporating section.

In addition, the loop heat transfer device of this invention is preferably does not have a fluid reservoir, a complex start unit, a control unit, a pump or other units typically required in starting and operating a conventional heat pipe. This device is also automatically restarted without failure when thermal load acts on the device after a complete stop of the device. Therefore, the loop heat transfer device of this invention may be preferably used in portable air conditioners, portable dehumidifiers, portable dryers, devices for recovering energy from exhaust gas, and white smoke preventing devices.

Although the preferred embodiments of the present invention have been disclosed for illustrative purposes, those skilled in the art will appreciate that various modifications, additions and substitutions are possible, without departing from the scope and spirit of the invention as disclosed in the accompanying claims.